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(71) Applicant: SHOWA ALUMINUM CORPORATION
Sakaishi, Osaka (JP)

(72) Inventors:
- Nakamura, Junpei,
c/o Showa Aluminum Corporation
Sakaishi, Osaka (JP)
- Yamazaki, Keiji,
c/o Showa Aluminum Corporation
Sakaishi, Osaka (JP)
- Higo, Yutaka,
c/o Showa Aluminum Corporation
Sakaishi, Osaka (JP)

(74) Representative: Paul, Dieter-Alfred, Dipl.-Ing. et al
Fichtestrasse 18
41464 Neuss (DE)

(54) Refrigeration cycle system

(57) The present invention relates to a refrigerant cycle system including a compressor (1), a condenser (2), a depressurizing means (4) and a evaporator (3) which are connected in series with each other to form a refrigerant circulation circuit. The refrigerant cycle system includes a heat exchanging portion (11). The heat exchanging portion (11) exchanges heat between a part of or all of the refrigerant passing through a refrigerant passage (5) from the compressor (1) to a depressurizing means (4) by way of the condenser (2) and a part of or all of the refrigerant passing through a refrigerant passage (6) from the depressurizing means (4) to the compressor (1) by way of the evaporator (3).

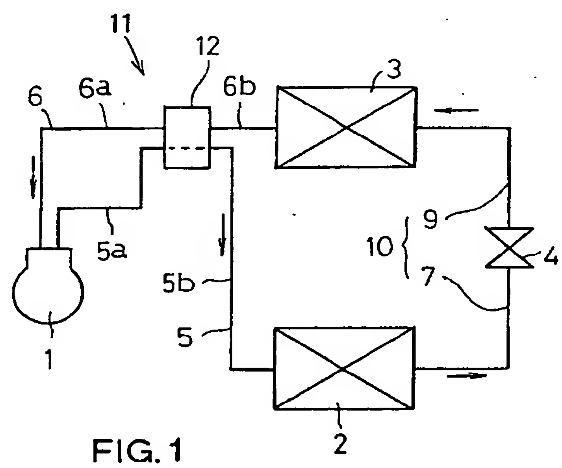

FIG. 1

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Description

FIELD OF THE INVENTION

This invention relates to a refrigeration cycle system for use in a refrigeration device such as an automobile air conditioner, a room air conditioner, or the like.

BACKGROUND OF THE INVENTION

A conventional refrigeration cycle system is shown in FIG. 14. The refrigeration cycle system comprises a compressor 51, a condenser 53 connected to the compressor 51 at the refrigerant output side thereof by way of a refrigerant passage 52 and an evaporator 55 connected to the compressor 51 at the refrigerant input side thereof by way of a refrigerant passage 54. The output side of the condenser 53 and the input side of the evaporator 55 are connected by refrigerant passages 57, 59 and an expansion valve 56, or a capillary tube, or the like, as a decompression means intervening therebetween. In this refrigeration cycle system, as indicated by arrows in FIG. 14, the following cycle is repeated. A high pressure and high temperature gaseous refrigerant from the compressor 51 is condensed in the condenser 53 to become a high pressure and high temperature liquid refrigerant. The high pressure and high temperature liquid refrigerant is passed through the expansion valve 56 to become a low pressure and low temperature liquid refrigerant. Then, the low pressure and low temperature liquid refrigerant is evaporated in the evaporator 55 to become a low pressure and low temperature gaseous refrigerant. Then, the low pressure and low temperature gaseous refrigerant is returned to the compressor 51.

The above mentioned conventional refrigeration cycle system is designed to superheat the refrigerant so as to increase the refrigeration effect and thus improve the performance of the refrigeration cycle. Specifically, the evaporator 55 is designed to have a superheating portion in the refrigerant passage near the outlet such that almost only gasified refrigerant passes through the superheating portion. Therefore, a liquid refrigerant is prevented from returning to the compressor 51 from the evaporator 55.

The above mentioned conventional refrigeration cycle system is also designed to supercool (subcool) the refrigerant so as to improve the performance of the refrigeration cycle. Specifically, the condenser 53 is designed to have a supercooling portion (subcooling portion) in the refrigerant passage near the outlet such that only liquefied refrigerant passes through the supercooling portion.

However, in the evaporator 55 having the superheating portion as mentioned above, there has been a problem in that the heat transfer rate between the refrigerant and the evaporator 55 is decreased which lowers the heat exchanging performance thereof, as compared to an evaporator not having a superheating portion therein.

More specifically, providing a superheating portion in the evaporator 55 means providing a flow area for a gasified refrigerant in the evaporator 55. Therefore, in the evaporator 55, refrigerant flowing through a refrigerant flow area, except for the superheating portion, is in a liquid state or in a sprayed state. The refrigerant which is in a liquid state or sprayed state will be gasified, thereby increasing the heat transfer rate. However, the refrigerant flowing through the superheating portion is already in a gasified state, thus the heat transfer rate in the superheating portion is low. As a result, when observed as a whole, the heat transfer rate between the refrigerant and the evaporator 55 is decreased and thus the heat exchange performance of the evaporator 55 is low.

An experiment was done so as to compare the heat transfer rate, i.e., the heat exchange performance, of two similar evaporators, one superheated by 5 degrees, the other not superheated. The result was that the heat exchange performance of the former was lower than the latter by 3 to 7 %.

Therefore, an evaporator which is superheated is inferior in heat exchange performance as compared to an evaporator having no superheating, provided that both evaporators are the same in size. Thus, in order to demonstrate the same heat exchange performance in both evaporators, one having superheating and the other having no superheating, the evaporator having superheating must be larger in size than the evaporator having no superheating because the evaporator having superheating has a superheating portion.

Furthermore, in the evaporator 55 which is superheated, the pressure loss of the refrigerant passing through the evaporator 55 is larger than that of the refrigerant passing through an evaporator having no superheating, thereby increasing the pressure loss of the refrigerant in the whole refrigerant cycle. In detail, the refrigerant passing through the superheating portion is in a gaseous state and has a large specific volume as compared to the refrigerant in a liquid state or in sprayed state (i.e., in a gas and liquid mixed state). As a result, because the specific volume in the superheating portion is large and because the refrigerant passages of the evaporator are narrow, the pressure loss of the refrigerant passing through the evaporator becomes larger.

An experiment was conducted to compare the pressure loss of the refrigerant passing through two similar evaporators, one being superheated by 5 degrees and the other not being superheated. The result of this test was that the heat pressure loss of the refrigerant passing through the former was higher than that of the refrigerant passing through the latter by 15 to 35%.

As shown in FIG. 15, an accumulator 60 may be provided within a refrigerant-passage connecting the evaporator 55 and the compressor 51 so as to decrease or delete the effect of the superheating portion of the evaporator 55. In this system, the liquid refrigerant which remains unevaporated in the evaporator 55 will

be captured by the accumulator 60. In this manner, the liquid refrigerant is prevented from returning to the compressor 51, and the heat transfer ratio between the evaporator 55 and the refrigerant, i.e., the performance of the evaporator 55, can be improved. Furthermore, the evaporator 55 can thus be smaller in size and the pressure loss of the refrigerant passing through the evaporator 55 can be decreased.

However, the accumulator 60 merely captures the liquid refrigerant which remains unevaporated in the evaporator 55. Thus, the refrigeration cycle system can only have a small number of degrees of superheating, or even no degrees of superheating. As a result, the refrigeration effect will not be improved with such superheating.

In other words, if superheating is effected in the evaporator so as to increase the refrigeration effect, a deterioration of the heat exchange performance, an increase in the size, and an increase in the refrigerant pressure loss will be caused. On the other hand, if an improvement of the heat exchange performance, a decrease in size, and a decrease in refrigerant pressure loss are attempted, an enhanced refrigeration effect due to the superheating will not be caused.

As for the condenser 53, having a supercooling portion, the heat transfer rate between the refrigerant and the condenser 53 is decreased, which deteriorates the heat exchange performance thereof, as compared to a condenser having no supercooling portion therein. More specifically, providing a supercooling portion in the condenser 53 means providing a refrigerant flow area for a liquefied refrigerant in the condenser 53. Therefore, in the condenser 53 a refrigerant flowing through a refrigerant flow area, except for the supercooling portion, is in a gaseous state or in a sprayed state. The refrigerant in a gaseous state or in a sprayed state proceeds to be liquefied, thereby increasing the heat transfer rate. On the other hand, the refrigerant flowing through the supercooling portion is in a liquefied state, and thus the heat transfer rate in the supercooling portion deteriorates. As a result, when observed as a whole, the heat transfer ratio between the refrigerant and the condenser 53 is decreased and thus the heat exchange performance of the condenser 53 deteriorates.

Therefore, the condenser having such supercooling is inferior in heat exchange performance to a condenser having no supercooling, provided that both condensers are the same in size. Thus, in order to demonstrate the same heat exchange performance in both condensers, one having supercooling and the other not having supercooling, the condenser having supercooling must be larger in size than the condenser not having supercooling because of the supercooling portion.

In other words, if supercooling is effected in the condenser so as to improve the performance of the refrigeration cycle, a deterioration in the heat exchange performance and an increase in size is caused. On the other hand, if an improvement in the heat exchange performance of the condenser and a decrease in size are attempted, an enhancement of the refrigeration performance by supercooling can not be caused.

Especially in a condenser, though there has been a demand for miniaturization and especially to be air cooled, it is very difficult to have an effective supercooling portion. Therefore, there has been the problem related to a refrigerant cycle of how to improve the performance thereof.

SUMMARY OF THE INVENTION

The present invention overcomes, among other things, the problems mentioned above. It is an object of the invention to provide a refrigeration cycle system in which the superheating degree and the supercooling degree can effectively become large and thus achieve an enhancement in the refrigerant effects and an improvement in the performance of the refrigerant cycle.

It is also an object of the invention to provide an improved refrigeration cycle system in which an evaporator and a condenser can be compact and can improve heat exchanging performance thereof.

It is another object of the invention to provide an improved refrigeration cycle system in which a pressure loss of a refrigerant passing through an evaporator can be decreased.

According to a first aspect of the present invention, a refrigerant cycle system comprises a refrigerant circulation circuit including a compressor, a condenser,an evaporator and a depressurizing means and a heat exchanging portion. The heat exchanging portion exchanges heat between at least a portion of refrigerant flowing from the compressor to the depressurizing means and at least a portion of refrigerant flowing from the depressurizing means to the compressor.

Because the refrigerant cycle system includes a heat exchanging portion for exchanging heat between at least a portion of refrigerant flowing from the compressor to the depressurizing means and at least a portion of refrigerant flowing from the depressurizing means to the compressor. Therefore, the refrigerant returning to the compressor can be superheated and the refrigerant flowing toward the depressurizing means can be supercooled. As a result, the refrigerant effect can be increased and the performance as the refrigerant cycle can be improved.

Notably, each refrigerant is either superheated or supercooled by exchanging heat between a low temperature refrigerant and a high temperature refrigerant which are greatly different in temperature. Therefore, each refrigerant can be effectively superheated or supercooled and, thus, can greatly improve the performance of the refrigerant cycle as compared with a conventional refrigerant cycle in which each refrigerant is separately superheated or supercooled by air at room temperature.

Further, the superheat portion in the evaporator can be decreased or omitted because the refrigerant return-

ing to the compressor is superheated by exchanging heat in the heat exchanging portion. Therefore, the evaporator can be compact and superior in heat exchange performance, and thus the pressure loss of the refrigerant passing through the evaporator can be deceased.

The refrigerant flowing toward the expansion valve can be largely supercooled because the refrigerant is supercooled by exchanging heat in the heat exchanging portion. Therefore, the dry degree of the liquefied refrigerant passed through the expansion valve can be effectively lowered. Further, the pressure loss of the refrigerant passing through the evaporator can be effectively decreased. Furthermore, the heat exchanging performance of the evaporator can be effectively improved.

The above and other objects and features of the invention will be apparent from the following detailed description of the invention with reference to the attached drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 illustrates a first embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 2 is an inner side view of an accumulator equipped in the above refrigerant cycle system.

FIG. 3 illustrates a second embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 4 illustrates a third embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 5 illustrates a fourth embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 6 is an inner side view of a heat exchanging portion integrating a liquid-receiver with an accumulator of the fourth embodiment.

FIG. 7 illustrates a filth embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 8A and 8B illustrate a sixth embodiment of a refrigerant cycle system according to the present invention, wherein FIG. 8A illustrates a refrigerant circuit and FIG. 8B is an explanatory view of the heat exchanging portion.

FIG. 9 illustrates a seventh embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 10A and 10B illustrate an evaporator and a heat exchanging portion of the seventh embodiment, wherein FIG. 10A is an inner front view thereof and FIG. 10B is an inner plan view thereof.

FIG. 11 illustrates an eighth embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 12 illustrates a ninth embodiment of a refrigerant circuit of a refrigerant cycle system according to the present invention.

FIG. 13A and 13B illustrate a tenth embodiment of a refrigerant cycle system according to the present invention, wherein FIG. 13A illustrates a refrigerant circuit of the embodiment ad FIG. 13B is an explanatory inner view of a valve device.

FIG. 14 illustrates a refrigerant circuit of a conventional refrigerant cycle system.

FIG. 15 illustrates a refrigerant circuit equipped with an accumulator in the conventional refrigerant cycle system.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Preferred embodiments of the present invention will now be described, in detail, with reference to the accompanying drawings.

First embodiment:

In the first embodiment of the refrigerant cycle system shown in FIG. 1, the numeral 1 denotes a compressor, the numeral 2 denotes a condenser, the numeral 3 denotes an evaporator, the numeral 4 denotes an expansion valve as a depressurizing means and the numeral 11 denotes a heat exchanging portion.

The condenser 2 is connected to the compressor 1 at the refrigerant output side thereof by way of refrigerant passages 5 (5a and 5b). The evaporator 3 is connected to the compressor 1 at the refrigerant input side thereof by way of refrigerant passages 6 (6a and 6b). The output side of the condenser 2 and the input side of the evaporator 3 are connected by refrigerant passages 10 (7 and 9) with an expansion valve 4 intervened therein to form a refrigerant circuit. As a decompression means, a capillary tube, an orifice tube, or the like, may be used.

In this embodiment, the heat exchanging portion 11 exchanges heat between the refrigerant passing through the refrigerant passage 5 connecting the compressor 1 to the evaporator 3 and the refrigerant passing through the refrigerant passage 6 connecting the evaporator 3 to the compressor 1. Further, in this embodiment, an accumulator 12 is provided within the refrigerant passage 6 (6a and 6b) connecting the compressor 1 and the evaporator 3. This accumulator 12 is used as the heat exchanging portion 11 and exchanges heat between the refrigerant accumulated in the accumulator 12 and the refrigerant passing through the refrigerant passage 5 connecting the compressor 1 to the condenser 2.

In order to achieve the above mentioned heat exchange, in this embodiment, the accumulator 12 is configured as shown in FIG. 2. The accumulator 12 includes a liquid accumulating container 13 which is, at the upper portion, provided with a first refrigerant inlet port 14 and a first refrigerant outlet port 15. The first

refrigerant outlet port 15 is connected in fluid communication with one end of a pipe 16 extended into the container 13 with the other end of the pipe 16 opening proximate at the uppermost portion in the container 13, such that as the refrigerant is introduced in the container 13 through the first refrigerant inlet port 14, the liquid refrigerant can be accumulated in the lower portion of the container 13 and the gaseous refrigerant can be led to the outside through the pipe 16 and the first refrigerant outlet port 15. So far, the above mentioned construction of the accumulator 12 is similar to a conventional accumulator. However, in the accumulator 12 the container 13 is further provided with a second refrigerant inlet port 17 and a second refrigerant outlet port 19. Both the ports 17 and 19 are connected in fluid communication via a high heat conductance heat exchanging pipe 20 extending through the container 13 so that heat exchange between the refrigerant in the heat exchanging pipe 20 and the refrigerant accumulated in the container 13 can be achieved.

In the accumulator 12, the first refrigerant inlet port 14 is connected to a pipe constituting the refrigerant passage 6b from the evaporator 3 and the first refrigerant outlet port 15 is connected to a pipe constituting the refrigerant passage 6a toward the compressor 1, and further the second refrigerant inlet port 17 is connected to a pipe constituting the refrigerant passage 5a from the compressor 1 and the second refrigerant outlet port 19 is connected to a pipe constituting the refrigerant passage 5b toward the condenser 2. Thus, the accumulator 12 is built in the refrigerant cycle.

In the refrigerant cycle system as constructed above, during the operation, a low temperature refrigerant from the evaporator 3 is passed through the liquid accumulating container 13 of the accumulator 12 and then returned to the compressor 1. On the other hand, a high temperature refrigerant from the compressor 1 is passed through the heat exchanging pipe 20 provided in the liquid accumulating container 13 of the accumulator 12 and then introduced to the condenser 2. Thus, the low temperature refrigerant accumulated in the container 13 is heated by the high temperature refrigerant in the heat exchanging pipe 20. Therefore, the refrigerant returning to the compressor 1 is superheated and completely gasified. The high temperature refrigerant in the heat exchanging pipe 20 is cooled by the low temperature refrigerant in the accumulating container 13. Therefore, the refrigerant sent to the expansion valve 4 is supercooled and completely liquefied. By the heat exchange mentioned above, the gaseous refrigerant returning to the compressor 1 can be superheated and the liquified refrigerant sent to the expansion valve 4 can be supercooled, in such a manner that the refrigerant effect is increased and the performance as the refrigerant cycle is improved.

Notably, each refrigerant is superheated or supercooled by exchanging heat between the low temperature refrigerant and the high temperature refrigerant which are greatly different in temperature. Therefore, each refrigerant can be effectively superheated or supercooled as compared to the conventional refrigerant cycle in which each refrigerant is superheated or supercooled by air at room temperature, thus the performance of the refrigerant cycle can be greatly improved.

Further, the superheat portion in the evaporator 3 can be decreased or omitted because the refrigerant returning to the compressor 1 is superheated by exchanging heat in the heat exchanging portion 11. Therefore, the evaporator 3 can be compact and superior in heat exchange performance, and the pressure loss of the refrigerant passing through the evaporator 3 can be decreased.

Furthermore, the refrigerant sent to the expansion valve 4 can be largely supercooled because the refrigerant is supercooled by exchanging heat in the heat exchanging portion 11. Therefore, the dry degree of the liquefied refrigerant passed through the expansion valve 4 can be effectively lowered. Further, the pressure loss of the refrigerant passing through the evaporator 3 can be effectively decreased and the heat exchanging performance of the evaporator 3 can also be effectively improved.

Because the accumulator 12 is modified to include the heat exchanging portion 11 as mentioned above, a large refrigerating ability of the accumulator 12, not previously contemplated, can be effectively provided to supercool the refrigerant flowing toward the expansion valve 4, and thus energy can be effectively utilized. Further, the original gas-liquid separating function of the accumulator 12 can be improved and thus the liquid refrigerant is effectively prevented from returning to the compressor 1.

Second embodiment:

FIG. 3 illustrates a second embodiment of the refrigerant cycle system according to the present invention. The heat exchanging portion 11 exchanges heat between the refrigerant passing through the refrigerant passage 7 (7a and 7b) connecting the condenser 2 to the expansion valve 4 and the refrigerant passing through the refrigerant passage 6 (6a and 6b) connecting the evaporator 3 to the compressor 1. In order to exchange heat as mentioned above, in the accumulator 12 which has the same structure of the accumulator of the first embodiment and has the second refrigerant inlet port 17 connected to the pipe constituting the refrigerant passage 7a from the condenser 2 and has the second refrigerant outlet port 19 connected to the pipe constituting the refrigerant passage 7b toward the expansion valve 4. Thus, the accumulator 12 is built in the refrigerant cycle.

In this second embodiment of the refrigeration cycle system, similar functions and results as in the first embodiment can be achieved. Further, notably, the supercooling portion in the condenser 2 can be decreased or omitted. Therefore, the condenser 2 can

be compact and superior in heat exchange performance. In particular, according to this embodiment, even if the supercooling portion of the condenser 2 which passes through the liquefied refrigerant is decreased or omitted, the refrigerant from the condenser 2 is supercooled by passing through the heat exchanging portion 11. Thus, the supercooling portion in the condenser 2 can be decreased or omitted. Therefore, the condenser 2 can be effectively compact and superior in its heat exchange performance.

Furthermore, especially in comparison with the first embodiment, the refrigerant condensing ability of the condenser 2 can be improved. In particular, in the first embodiment, the condensing ability of the condenser 2 somewhat deteriorates because the refrigerant cooled in the heat exchanging portion 11 is subjected to be condensed in the condenser 2. On the contrary, in the second embodiment, the condenser 2 can maintain high condensing ability thereof because the high temperature refrigerant from the compressor 1 is directly fed into the condenser 2 and condensed therein.

Third embodiment:

FIG. 4 illustrates a third embodiment of the refrigerant cycle system. In the third embodiment, similar to the second embodiment, the heat exchanging portion 11 exchanges heat between the refrigerant passing through the refrigerant passage 7 (7a, 7b and 7C) connecting the condenser 2 to the expansion valve 4 and the refrigerant passing through the refrigerant passage 6 (6a and 6b) connecting the evaporator 3 to the compressor 1, and the accumulator 12 is used as a heat exchanging portion 11. Further, a liquid-receiver 32 is interposed in the refrigerant passage 7 (7c and 7a) connecting the condenser 2 to the expansion valve 4 so that the refrigerant from the condenser 2 exchanges heat in the heat exchanging portion 11 of the accumulator 12 after passing through the liquid-receiver 32 and then is send to the expansion valve 4.

In the third embodiment, alter the gaseous ingredient is removed from the refrigerant in the liquid-receiver 32, only the liquefied refrigerant is send to the accumulator 12 and then exchanges heat to be supercooled. The refrigerant flowing to the expansion valve 4 can be effectively supercooled in comparison with the case in which the liquefied refrigerant containing gaseous refrigerant exchanges heat in the heat exchanging portion 11.

Fourth embodiment:

FIG. 5 illustrates a fourth embodiment of the refrigerant cycle system. In the fourth embodiment, an accumulator and a liquid-receiver are integrated to form a heat exchanging portion 11. In detail, as shown in FIG. 6, the heat exchanging portion 11 includes a container 39 which is divided by a dividing wall 36 into an accumulator cell 12 and a liquid-receiving cell 32. The dividing wall 36 is equipped with fins 37 for promoting heat exchange between the accumulator cell 12 and the liquid-receiving cell 32.

The refrigerant passages 7a and 7b connecting the condenser 2 to the expansion valve 4 are connected to the liquid-receiving cell 32. The refrigerant passages 6a and 6b connecting the evaporator 3 to the compressor 1 are connected to the accumulator cell 12. In the fourth embodiment, a large quantity of a high temperature refrigerant and a low temperature refrigerant can very effectively exchange heat in the heat exchanging portion 11.

Fifth embodiment:

FIG. 7 illustrates a fifth embodiment of the refrigerant cycle system according to the present invention. This system is similar to the first embodiment but different from the first embodiment in that the heat exchanging portion 11 is not composed of an accumulator but of a heat exchange piping system in which heat exchanging is performed between heat exchange piping portions 27 and 29. As is apparent from the above, the heat exchanging portion 11 is not necessarily composed from an accumulator, this same concept can be applied to the second embodiment.

Sixth embodiment:

FIG. 8A illustrates a sixth embodiment of the refrigerant cycle system according to the present invention. In this system, the refrigerant passage 5 connecting the compressor 1 to the condenser 2 and the refrigerant passage 7 connecting the condenser 2 to the expansion valve 4 are bypassed by the bypass refrigerant passages 30, 30. The numeral 11 denotes a heat exchanging portion. The heat exchanging portion 11 exchanges heat between the refrigerant passing through the bypass refrigerant passage 30 and the refrigerant passing through the inner passage of the outlet side of the evaporator 3.

This heat exchanging portion 11 is, for example, constructed as follows. As shown in FIG. 8B, the heat exchanging portion 11 includes a final refrigerant passage 3b of the evaporator 3 connected to the outlet 3a, and an independent heat exchanging passage 31 adjacent to the final refrigerant passage 3b. In this heat exchanging portion 11, the heat exchanging passage 31 is interposed in fluid communication in the above mentioned bypass refrigerant passage 30. A liquid-receiver 32 is interposed in the refrigerant passage 7 between a position down stream of the juncture of the bypass refrigerant passage 30 and the refrigerant passage 7 and a position up stream from the expansion valve 4.

In this refrigerant cycle system, the high temperature refrigerant advancing toward the condenser 2 is divided into two refrigerant flow paths, a refrigerant flow path into the condenser 2 and a refrigerant flow path

into the bypass passage 30. The high temperature refrigerant introduced into the bypass passage 30 at the heat exchanging portion 11, exchanges heat with the low temperature refrigerant passing through the final refrigerant passage 3b of the evaporator 3. By this heat exchange, the low temperature refrigerant passing through the final refrigerant passage 3b of the evaporator 3 is heated and the refrigerant is progressively superheated. Further, by the above mentioned heat exchange, the high temperature refrigerant passing through the bypass passage 30 is cooled. After passing through the bypass passage 30, the refrigerant is merged with the refrigerant passed through the condenser 2, thereby enhancing the supercooling degree of the liquid refrigerant sent to the expansion valve 4. Notably, alter the merging, by passing through the liquid-receiver 32, the refrigerant passed through the bypass passage 30 and the refrigerant passed through the condenser 2 are mixed. Thus, the supercooling degree of the liquid refrigerant flowing toward the expansion valve 4 is effectively enhanced.

Further, the low temperature refrigerant passing through the final refrigerant passage 3b of the evaporator 3 is compulsively heated by the high temperature refrigerant as mentioned above. Thus, a sufficient superheat degree can be achieved and the superheating portion of the evaporator 3 can be effectively decreased. Therefore, the evaporator 3 can be small in size and can be enhanced in its heat exchanging performance. Further, the pressure loss of the refrigerant in the evaporator 3 can be decreased. Furthermore, the pressure loss of the refrigerant passing through the circuit can be decreased because an accumulator can be omitted.

Seventh embodiment:

FIG. 9 illustrates a seventh embodiment of the refrigerant cycle system according to the present invention. In the seventh embodiment, a heat exchanging portion 11 is equipped at the condenser 2. In detail, as shown in FIGS. 10A and 10B, this condenser 2 is so-called multi-flow or parallel-flow type heat exchanger having a plurality of tubes 41 whose ends are connected in fluid communication to a cylindrical hollow header 42. The numeral 43 denotes a fin. In this multi-flow type heat exchanger 2, the inside of the vertically disposed header 42 is divided by a dividing wall 44 having a high thermal conductivity into two chambers 45, 46. The chamber 45 is connected to the tubes 41, and the chamber 46 is not connected to the tubes 41. The chamber 45 connected to the tubes 41, as an essential part of the condenser, is connected to the refrigerant passages 7 toward the expansion valve 4. On the other hand, the chamber 46 not connected to the tubes 41, which functions as a part of an accumulator, is connected to the refrigerant passages 6a, 6b connecting the evaporator 3 to the compressor 1.

In this seventh embodiment, the high temperature refrigerant condensed in the tubes 41 is introduced into the chamber 45 of the header 42 connected to the tubes 41 and the low temperature refrigerant evaporated in the evaporator 3 is introduced into the chamber 46 of the header 42 not connected to the tubes 41. Both the refrigerants exchange heat through the dividing wall 44 such that the refrigerant toward the compressor 1 is superheated and the refrigerant toward the expansion valve 4 is supercooled.

In the seventh embodiment, an liquid-receiver may be interposed in a refrigerant passage 7 connecting the condenser 2 to the expansion valve 4.

Eighth embodiment:

FIG. 11 illustrates an eighth embodiment of the refrigerant cycle system according to the present invention. This refrigerant cycle system is especially useful for an automobile air conditioning system. In this refrigerant cycle system, a bypass passage 22 (22a and 22b) is equipped within the refrigerant passage 5 connecting the compressor 1 to the condenser 2. A heat exchanging portion 11 is provided so as to exchange heat between the refrigerant passing through the bypass passage 22 and the refrigerant passing through the refrigerant passage 6 connecting the evaporator 3 to the compressor 1. In order to establish the heat exchanging portion 11, an accumulator 12 having the same structure of the accumulator in the first embodiment is used. In this accumulator 12, the second refrigerant inlet port 17 is connected to the pipe constituting the bypass passage 22a from the compressor 1 and the second refrigerant outlet port 19 is connected to the pipe constituting the bypass passage 22b toward condenser 2. Further, the refrigerant inlet end portion of the bypass passage 22 is connected to the refrigerant passage 5 connecting the compressor 1 to the condenser 2 by way of a distributor 23. The distributor 23 changes the refrigerant flow such that the refrigerant from the compressor 1 is sent to the condenser 2 through the bypass passage 22 or the refrigerant from the compressor 1 is sent to the condenser 2, not through the bypass passage 22, but through the original refrigerant passage 5. A thermal sensor 24 is attached to a refrigerant outlet portion of the condenser 2 or nearby the outlet portion so as to detect the temperature of the refrigerant from the condenser 2. Alternatively, the thermal sensor 24 may be attached at the evaporator 3 side. The numeral 25 denotes a controller. The controller 25 is designed to output control signals to the distributor 23 for sending the refrigerant from the compressor 1 to the condenser 2 through the bypass refrigerant passage 22 based on the detected signals which are output from the thermal sensor 24 when the sensor 24 detects an overloaded temperature, i.e., a temperature higher than usual of the refrigerant from the condenser 2. The controller 25 may be composed of, for example, a micro computer.

In the eight embodiment of the refrigerant cycle system, when it is used in an automobile air condition-

ing system, superior results can be obtained.

In detail, in an automobile, the operating states thereof varies from an idling state to a low speed running state, or from the low speed running state to a high speed running state, or the like. Thus, in the condenser 2, the amount of the air flow which is heat exchanged with the refrigerant passing through the condenser 2 is changed depending on the operating state of the automobile. For example, when the automobile is in the idling state, the amount of the air flow passing through the condenser 2 is small. On the contrary, when the automobile is running at a high speed, the amount of the air flow passing through the condenser 2 is large. Therefore, when the automobile is running at a high speed, the condenser 2 actively exchanges heat, however, when the automobile is in the idling state, the performance of heat exchange with air in the condenser 2 deteriorates and thus the condenser 2 is overloaded. Under these circumstances, refrigerant cooling functions of the condenser 2 deteriorates and the supercooling degree of the liquid refrigerant flowing toward the expansion valve 4 becomes low. When the load of the condenser 2 becomes heavy, the heat exchanging performance of the evaporator 3 deteriorates. Thus, the ratio of the superheating degree of the gaseous refrigerant flowing to the compressor 1 becomes larger and the performance as a whole system deteriorates. As a result, the temperature in the car varies depending on the operating states of the car. Therefore, it is hard to realize a comfortable air conditioning environment.

In the eighth embodiment, when the condenser 2 is heavily loaded during an idling state, or the like, the situation is detected by the thermal sensor 24. The distributor 23 functions so as to send the refrigerant from the compressor 1 to the condenser 2 through the bypass passage 22 based on the control signals from the controller 25. The refrigerant flowing from the compressor 1 toward the condenser 2 exchanges heat with the refrigerant flowing from the evaporator 3 toward the compressor 1 by the accumulator 12. The low temperature gaseous refrigerant returning to the compressor 1 is heated by the high temperature gaseous refrigerant flowing toward the condenser 2 and is further superheated. The high temperature gaseous refrigerant flowing toward the condenser 2 is cooled by the low temperature gaseous refrigerant returning to the compressor 1, thus the liquid refrigerant sent to the expansion valve 4 is also further supercooled. Therefore, even if the load to the condenser 2 becomes large during an idling state or the like, a deterioration of the performance of the refrigerant cycle is restrained or prevented occurring. Thus, the temperature in the car become stable in spite of changes in the car operation state. Thus, a comfortable air conditioned environment is realized.

Ninth embodiment:

FIG. 12 illustrates a ninth embodiment of the refrigerant cycle system according to the present invention. In this refrigerant cycle system, a bypass passage 26 (26a and 26b) is equipped within the refrigerant passage 7 connecting the condenser 2 to the expansion valve 4. A heat exchanging portion 11 is provided so as to exchange heat between the refrigerant passing through the bypass passage 26 and the refrigerant passing through the refrigerant passage 6 connecting the evaporator 3 to the compressor 1. In order to establish the heat exchanging portion 11, an accumulator 12 having the same structure of the accumulator in the first embodiment is used. In the accumulator 12, the second refrigerant inlet port 17 is connected to the pipe constituting the bypass passage 26a from the condenser 2 and the second refrigerant outlet port 19 is connected to the pipe constituting the bypass passage 26b toward the expansion valve 4. Further, a distributor 23, a thermal sensor 24 and a controller 25 are provided in the same manner as per the third embodiment. In this ninth embodiment, effects which are the same as or superior to that of the eighth embodiment can be achieved.

Tenth embodiment:

FIG. 13A illustrates a tenth embodiment of the refrigerant cycle system according to the present invention. This refrigerant cycle system shown in FIG. 13A is similar to the sixth embodiment, but is different in that the bypass passage 30 is opened or closed by the valve device 34 shown in FIG. 13B. A thermal sensor 24 is attached to a refrigerant outlet portion of the evaporator 3 so as to detect the temperature of the refrigerant passing through the evaporator 3. Alternatively, the thermal sensor 24 may be attached to the condenser 2 side. The numeral 25 denotes a controller. The controller 25 is designed to output control signals to the valve device 34 for opening the bypass refrigerant passage 30, which is usually closed, based on detected signals which are output from the thermal sensor 24 when the sensor 24 detects an overloaded temperature, i.e., a temperature lower than usual, of the refrigerant of the evaporator 3. In the tenth embodiment of the refrigerant cycle system, when it is used in an automobile air conditioning system, superior results can be obtained as in the eight and ninth embodiments.

In detail, in an automobile, when the load to the condenser 2 becomes large during an idling state, or the like, the evaporator 3 becomes overloaded. Such a situation is detected by the thermal sensor 24 equipped at the evaporator 3 side, and then the bypass passage 30 is opened by the valve device 34 in accordance with the control signals from the controller 25. Therefore, a deterioration in the superheating of the gaseous refrigerant is restrained or prevented from occurring and a deterioration in the supercooling of the liquid refrigerant flowing toward the expansion valve 4 is restrained or prevented from occurring, thus a deterioration in the performance of the refrigerant cycle is restrained or prevented. Therefore, even if the load to the condenser 2 becomes large during an idling state, or the like, and

thus the evaporator 3 becomes overloaded, a deterioration in the performance of the refrigerant cycle is restrained or prevented. Thus, the temperature in the car becomes stable in spite of changes in the car's operation and a comfortable air conditioned environment can be realized.

As mentioned above, the refrigerant cycle system according to the present invention can include a heat exchanging portion for exchanging heat between at least a part of the refrigerant passing through a refrigerant passage from the compressor to a depressurizing means by way of the condenser and at least a part of refrigerant passing through a refrigerant passage from the depressurizing means to the compressor by way of the evaporator. Therefore, the superheating degree and supercooling degree can be efficiently enhanced and, thus, the performance of the refrigerant cycle can be improved. Further, the evaporator and the condenser can be compact and superior in heat exchange performance, and the pressure loss of the refrigerant passing through the evaporator can be decreased.

Although the invention has been described in connection with specific embodiments, the invention is not limited to such embodiments and as would be apparent to those skilled in the art, various substitutions and modifications within the scope and spirit of the invention are contemplated.

Claims

1. A refrigerant cycle system, comprising:

   a refrigerant circulation circuit including a compressor, a condenser, depressurizing means and an evaporator; and
   a heat exchanging portion;

   wherein said heat exchanging portion exchanges heat between at least a portion of refrigerant flowing from the compressor to the depressurizing means and at least a portion of refrigerant flowing from the depressurizing means to the compressor.

2. The refrigerant cycle system as recited in claim 1, wherein said heat exchanging portion exchanges heat between a refrigerant flowing from the compressor to the condenser and a refrigerant flowing from the evaporator to the compressor.

3. The refrigerant cycle system as recited in claim 1, wherein said heat exchanging portion exchanges heat between a refrigerant flowing from the condenser to the depressurizing means and a refrigerant flowing from the evaporator to the compressor.

4. The refrigerant cycle system as recited in claim 3, further including a liquid -receiver interposed within a refrigerant passage connecting the condenser to said heat exchanging portion.

5. The refrigerant cycle system as recited in claim 3, wherein said heat exchanging portion includes a container internally divided by a dividing wall into an accumulator cell and a liquid-receiving cell, the accumulator cell being interposed within a refrigerant passage connecting the condenser to the depressurizing means, the liquid-receiving cell being interposed within a refrigerant passage connecting the evaporator to the compressor, wherein the dividing wall is equipped with fins for promoting head exchange between refrigerant in the accumulator cell and refrigerant in the liquid-receiving cell.

6. The refrigerant cycle system as recited in claim 2, wherein said heat exchanging portion includes a pair of heat exchange pipings, one within a refrigerant passage connecting the evaporator to the compressor, the other within a refrigerant passage connecting the compressor to the condenser.

7. The refrigerant cycle system as recited in claim 1, further including a bypass refrigerant passage, the bypass refrigerant passage bypassing a refrigerant from the compressor to the outlet side of the condenser,

   wherein said heat exchanging portion exchanges heat between a refrigerant passing through the bypass refrigerant passage and a refrigerant passing through an outermost inner passage of the evaporator.

8. The refrigerant cycle system as recited in claim 7, wherein said heat exchanging portion includes a final refrigerant passage of the evaporator, the final refrigerant passage being connected to an outlet of the evaporator, and an independent heat exchanging passage being adjacent to the final refrigerant passage, the independent heat exchanging passage being interposed within the bypass refrigerant passage.

9. The refrigerant cycle system as recited in claim 8, further including a liquid receiver, the receiver being interposed within a refrigerant passage between a position down stream from a juncture of the bypass refrigerant passage and a refrigerant passage connecting the condenser to the depressurizing means and a position up streams from the depressurizing means.

10. The refrigerant cycle system as recited in claim 1,

   wherein the condenser includes a plurality of tubes whose ends are connected in fluid communication to a hollow header, the inside of the header being divided by a dividing wall having a high thermal conductivity into two chambers, one of the chambers being connected to the tubes and the

other being not connected to the tubes,

and wherein the chamber connected to the tubes is connected to a refrigerant passage to the depressurizing means, the chamber not connected to the tube is interposed within a refrigerant passage connecting the evaporator to the compressor.

11. The refrigerant cycle system as recited in claim 2, further including a bypass passage interposed within a refrigerant passage connecting the compressor to the condenser,

wherein said heat exchanging portion exchanges heat between a refrigerant passing through the bypass passage and a refrigerant passing through a refrigerant passage connecting the evaporator to the compressor.

12. The refrigerant cycle system as recited in claim 1, further including a bypass passage interposed within a refrigerant passage connecting the condenser to the depressurizing means,

wherein said heat exchanging portion exchanges heat between a refrigerant passing through the bypass passage and a refrigerant passing through a refrigerant passage connecting the evaporator to the compressor.

13. The refrigerant cycle system as recited in claim 1, further including a bypass refrigerant passage, a thermal sensor, a valve device and a controller, the bypass refrigerant passage bypassing refrigerant from the compressor to the outlet side of the condenser, the thermal sensor being attached to a refrigerant outlet portion of the evaporator so as to detect a temperature of refrigerant passing through the evaporator, the valve device being interposed within a refrigerant passage between the compressor and the condenser, the controller being designed to output control signals to the valve device for opening the bypass refrigerant passage based on a detected signal which is output from the thermal sensor when the sensor detects an overloaded temperature of refrigerant of the condenser,

wherein said heat exchanging portion exchanges heat between refrigerant passing through the bypass refrigerant passage and refrigerant passing through an outermost inner passage of the evaporator.

14. The refrigerant cycle system as recited in claim 1, 2, 3, 4, 11 or 12, wherein said heat exchanging portion includes an accumulator and a heat exchanging pipe provided in the accumulator,

wherein the accumulator includes a liquid accumulating container, said container being, at the upper portion, provided with a first refrigerant inlet port connected to a refrigerant passage from the evaporator and a first refrigerant outlet port connected to a refrigerant passage to the compressor, the first refrigerant outlet port being connected in fluid communication with one end of a pipe extends into the container with the other end of the pipe opened proximate the uppermost portion inside the container, the container being further provided with a second refrigerant inlet port connected to a refrigerant passage from the compressor and a second refrigerant outlet port connected to a refrigerant passage to the condenser,

and wherein both of the second outlet port and second inlet port are connected in fluid communication with a high heat conductance heat exchanging pipe so that heat exchange between a refrigerant in the heat exchanging pipe and a refrigerant accumulated in the container is achieved.

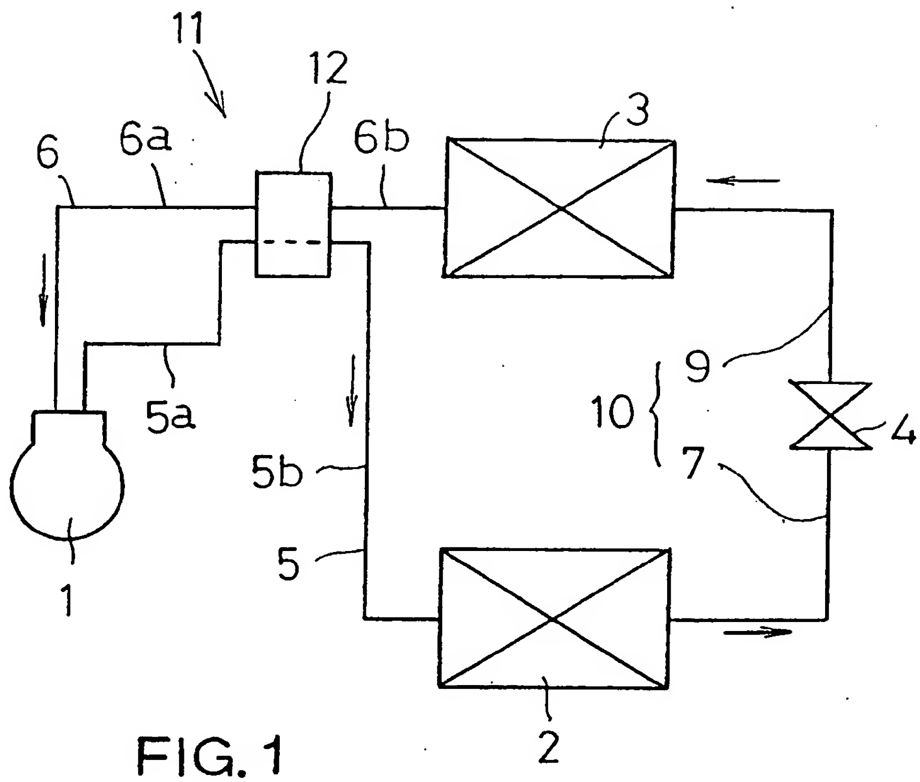

FIG. 1

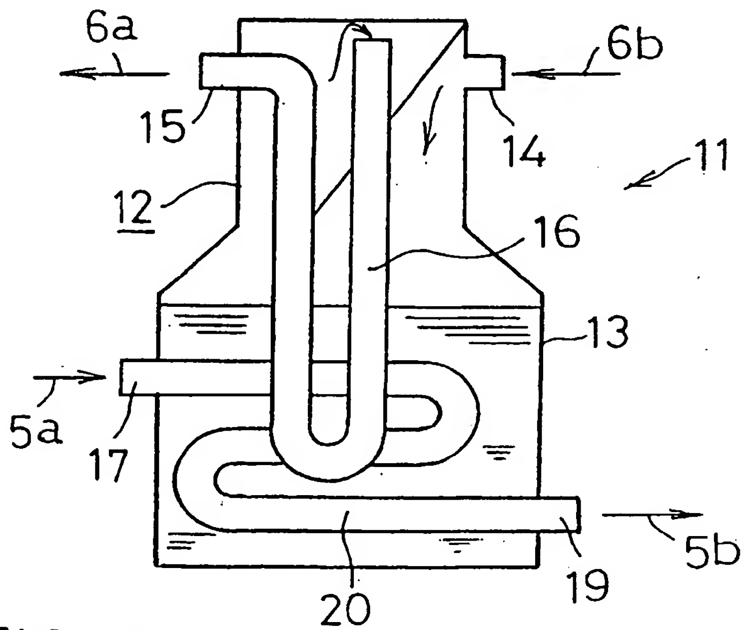

FIG. 2

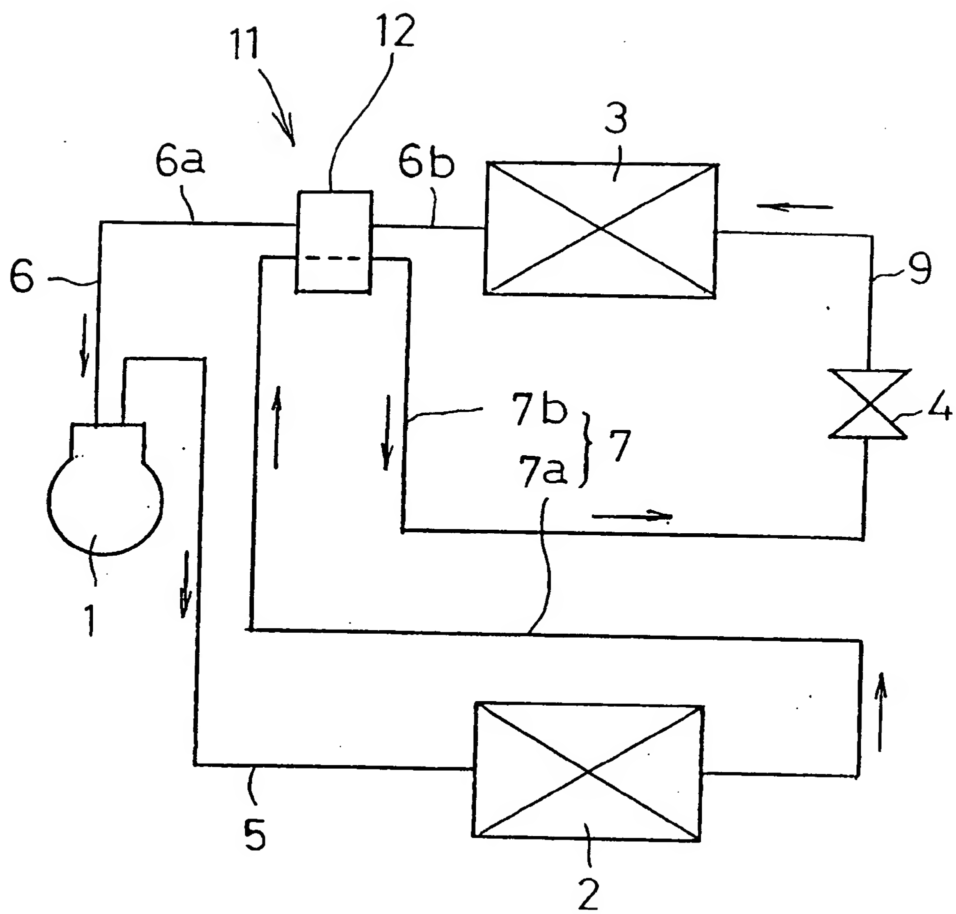

FIG. 3

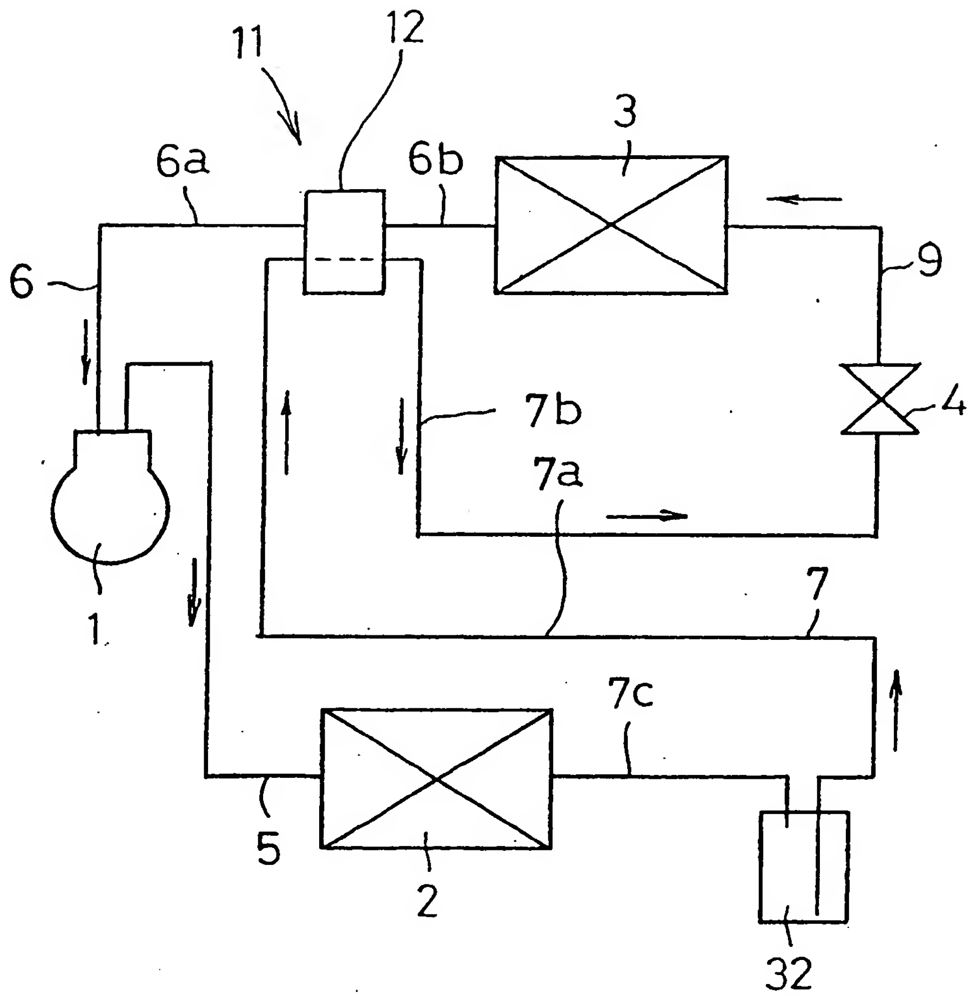

FIG. 4

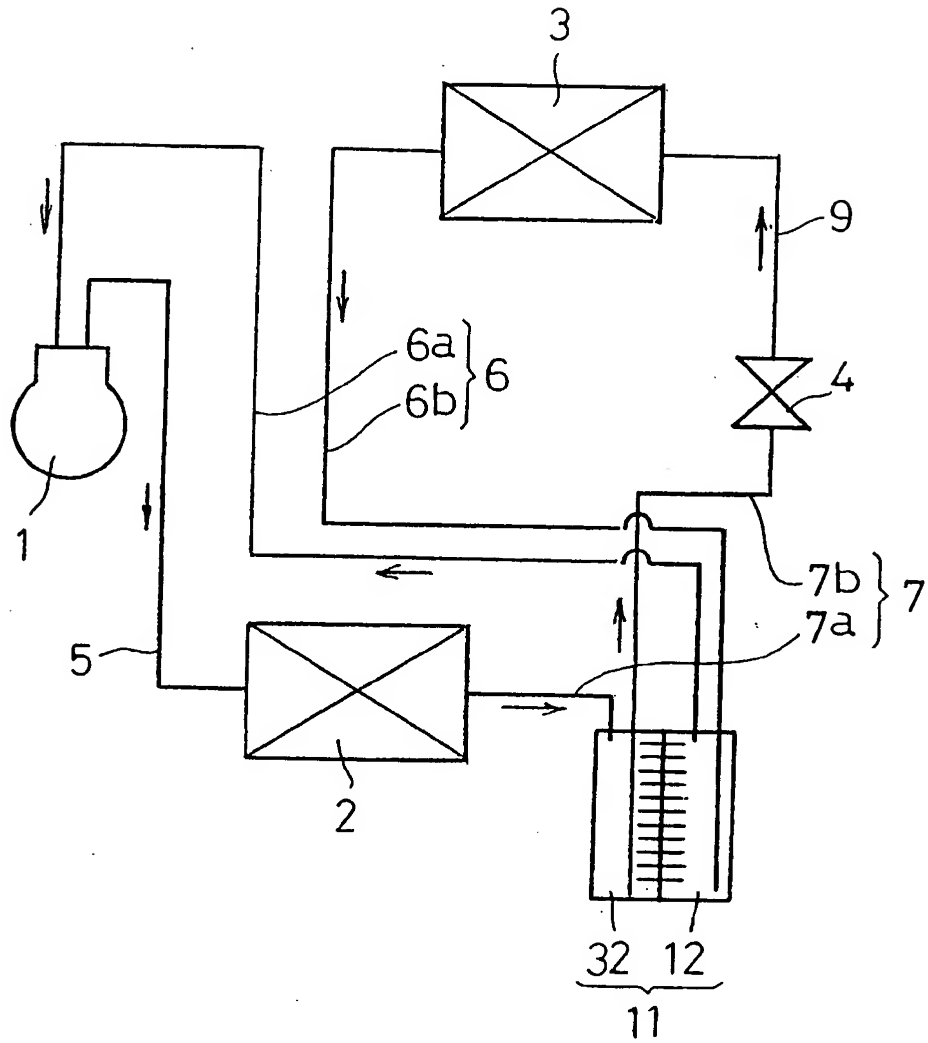

FIG. 5

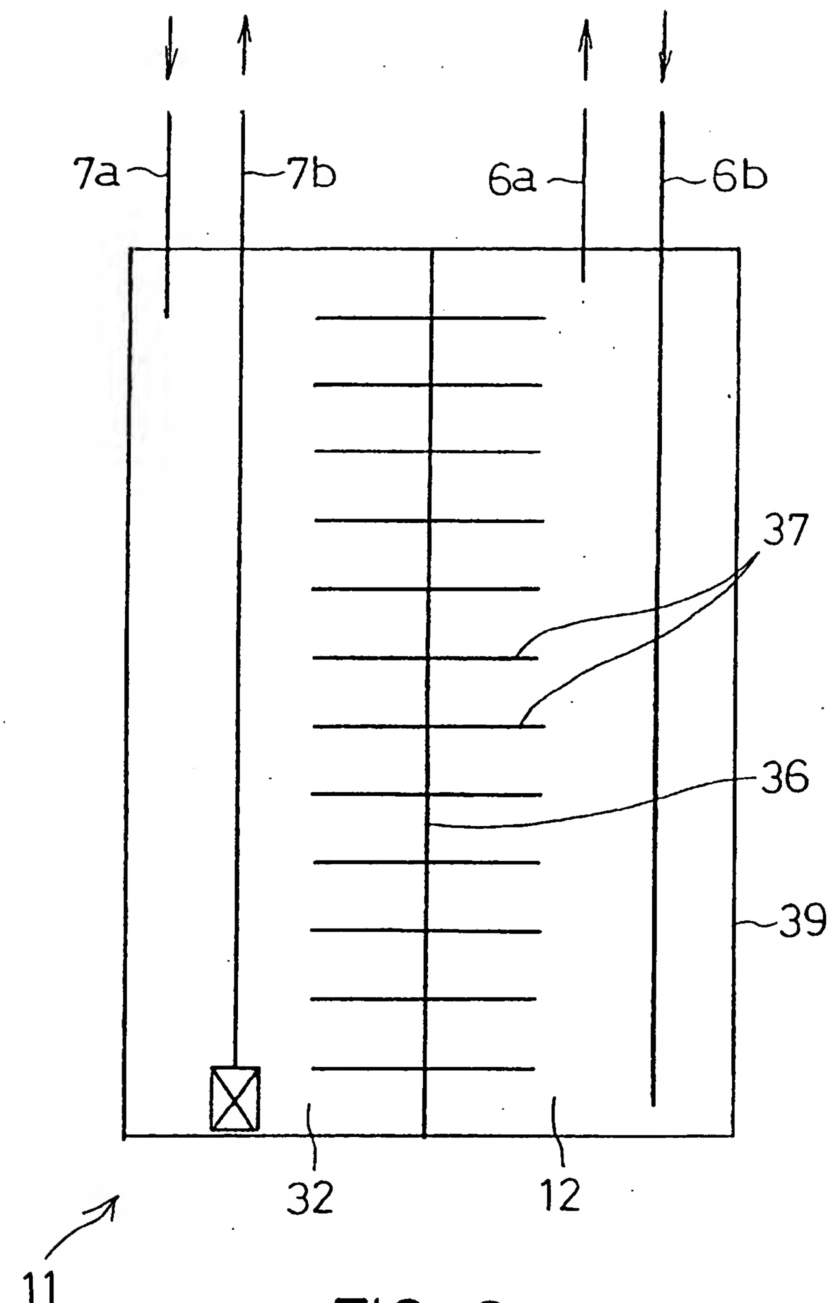

FIG. 6

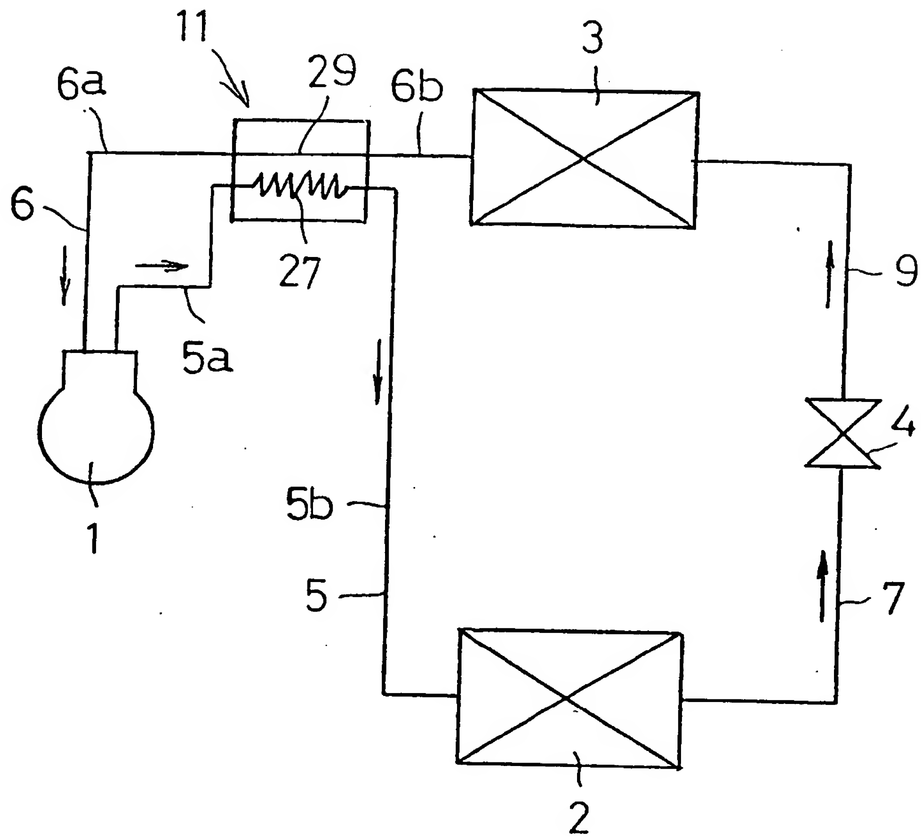

FIG. 7

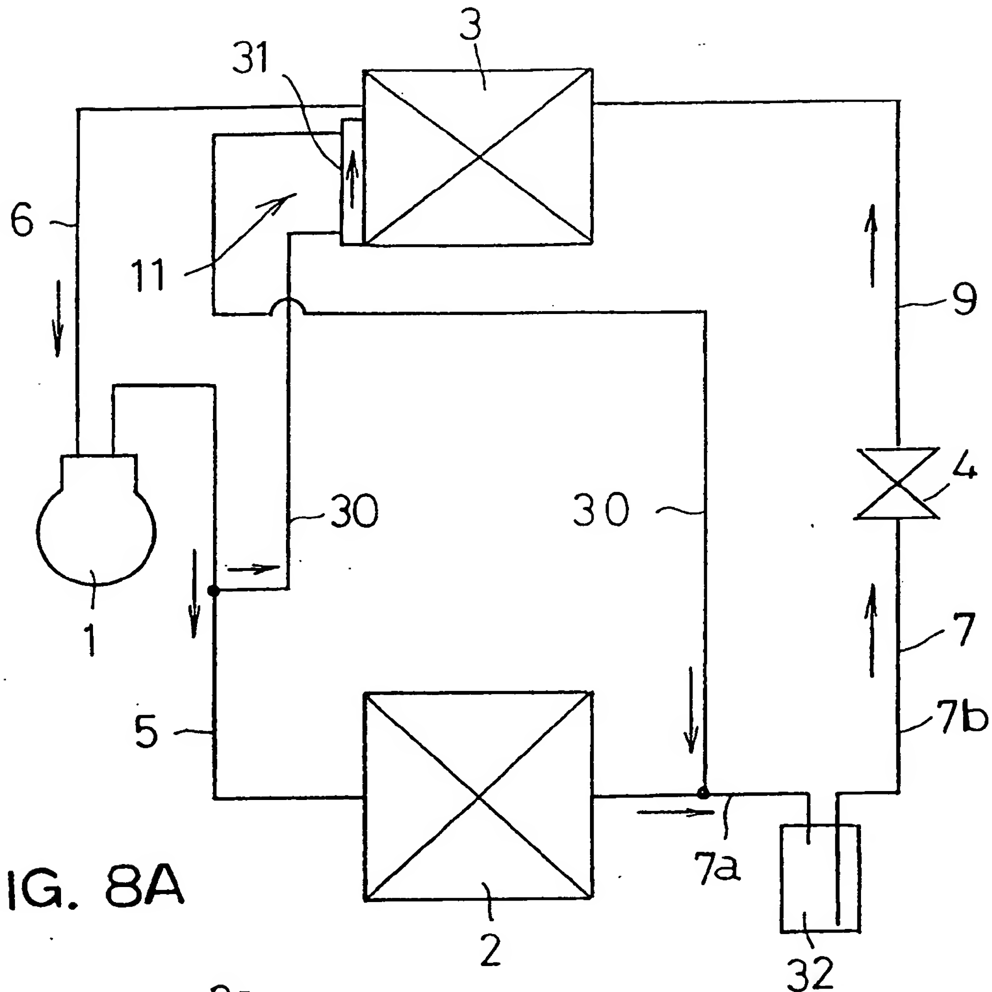

FIG. 8A

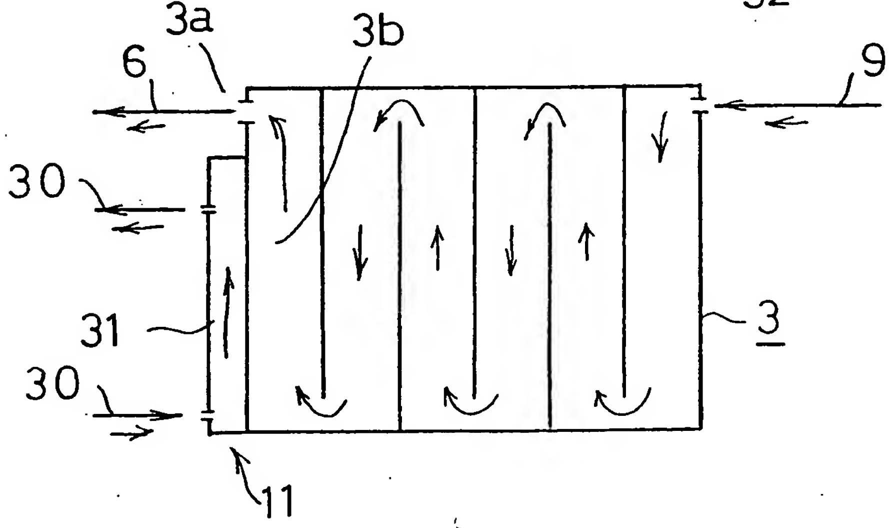

FIG. 8B

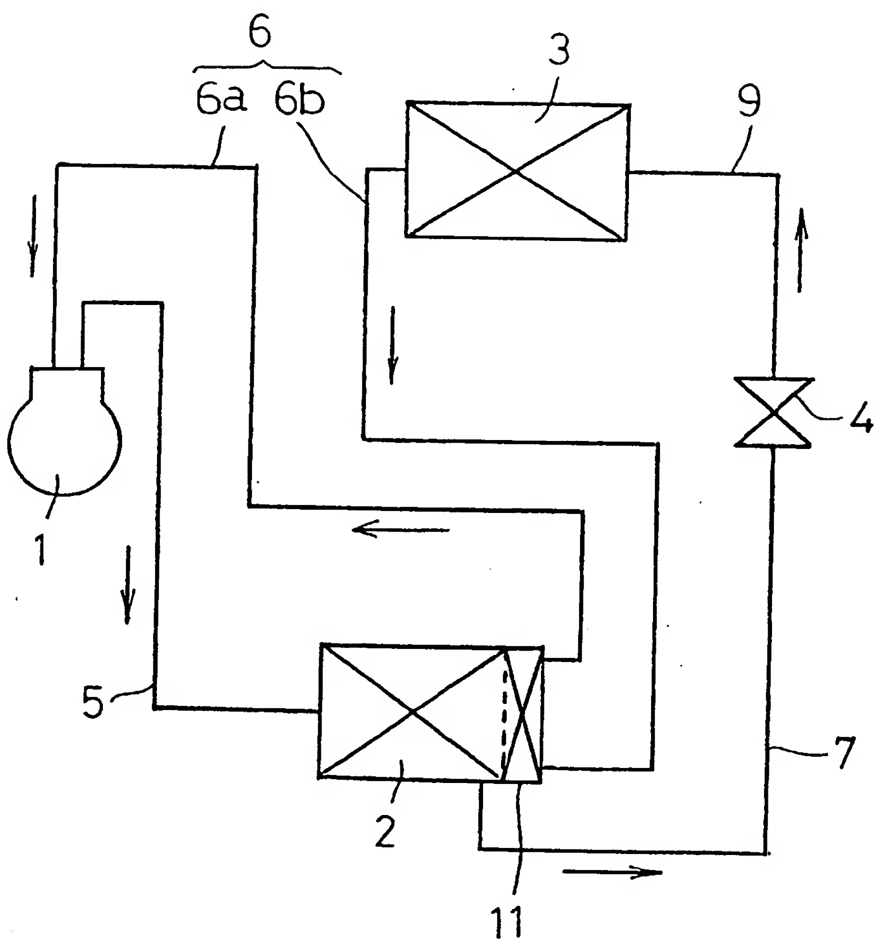

FIG. 9

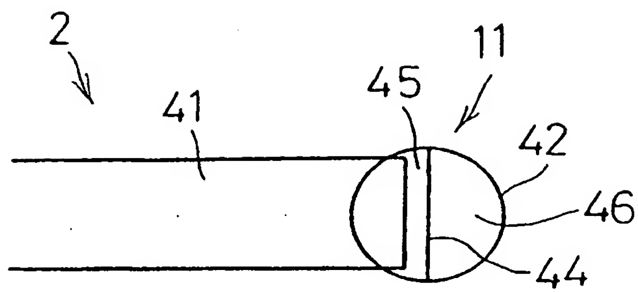

FIG. 10B

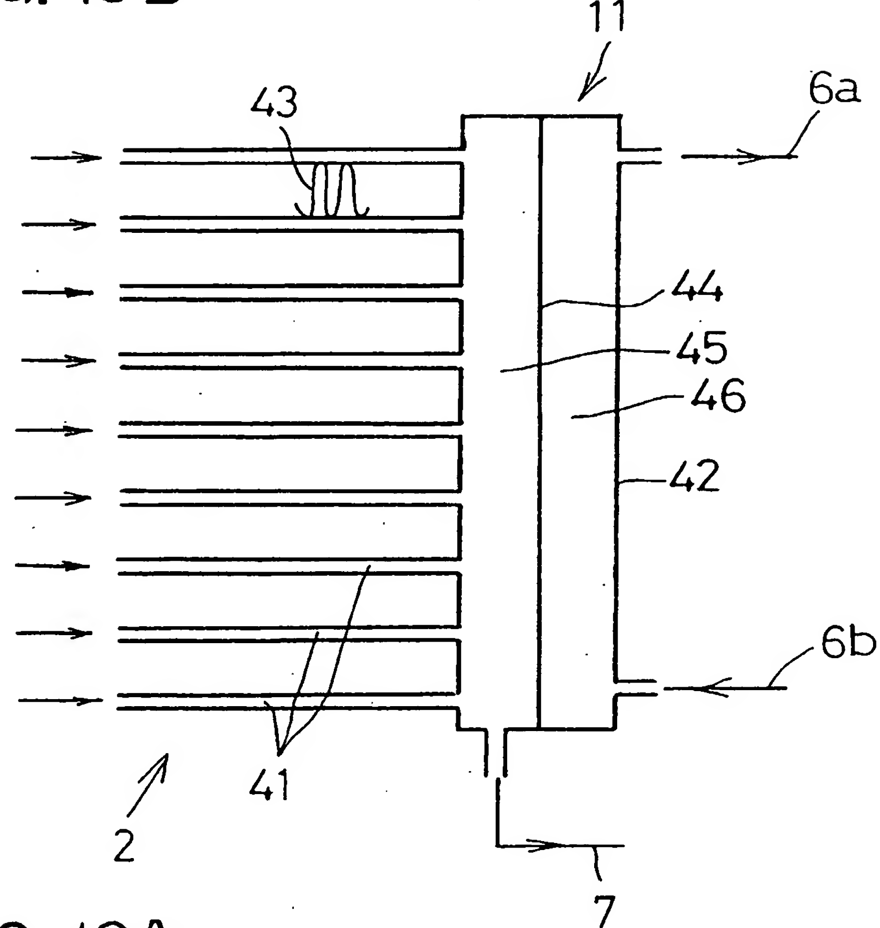

FIG. 10A

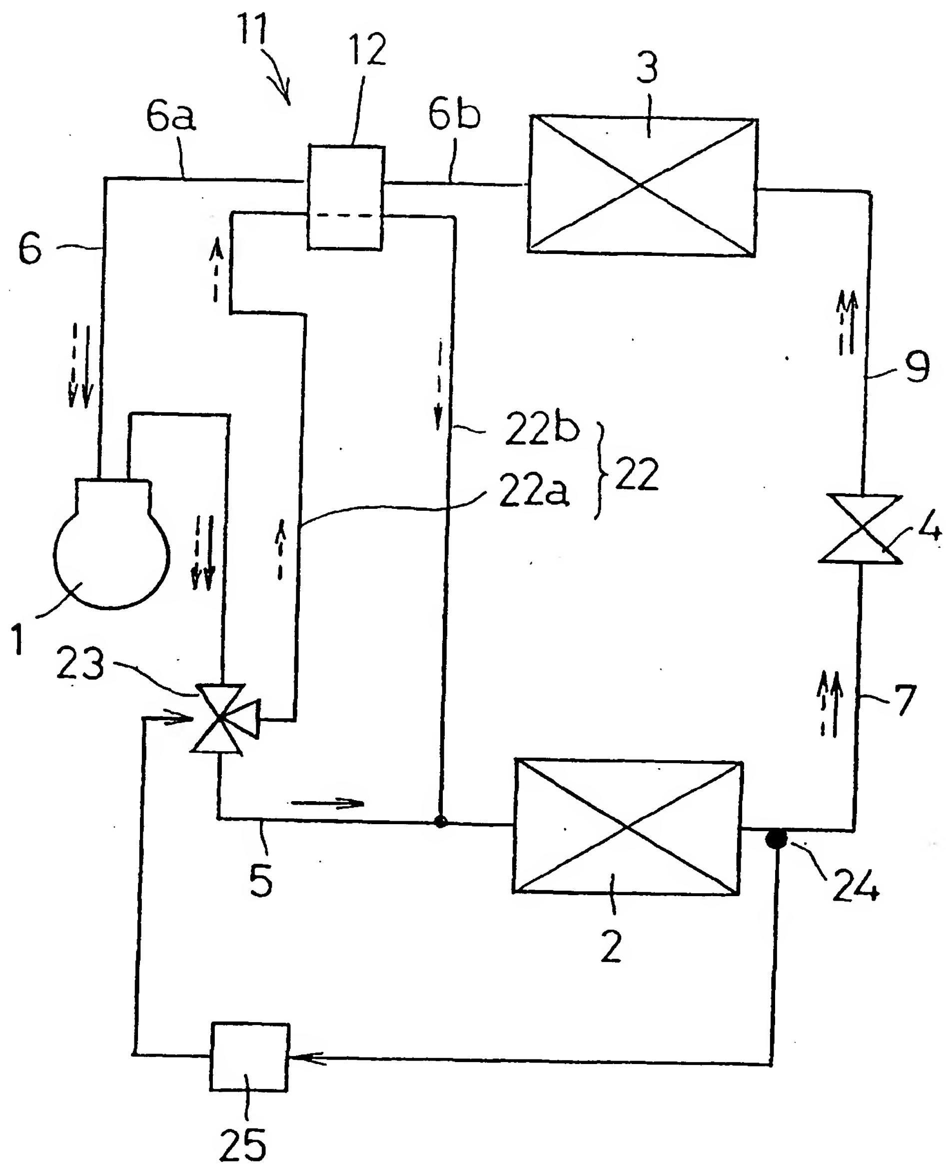

FIG. 11

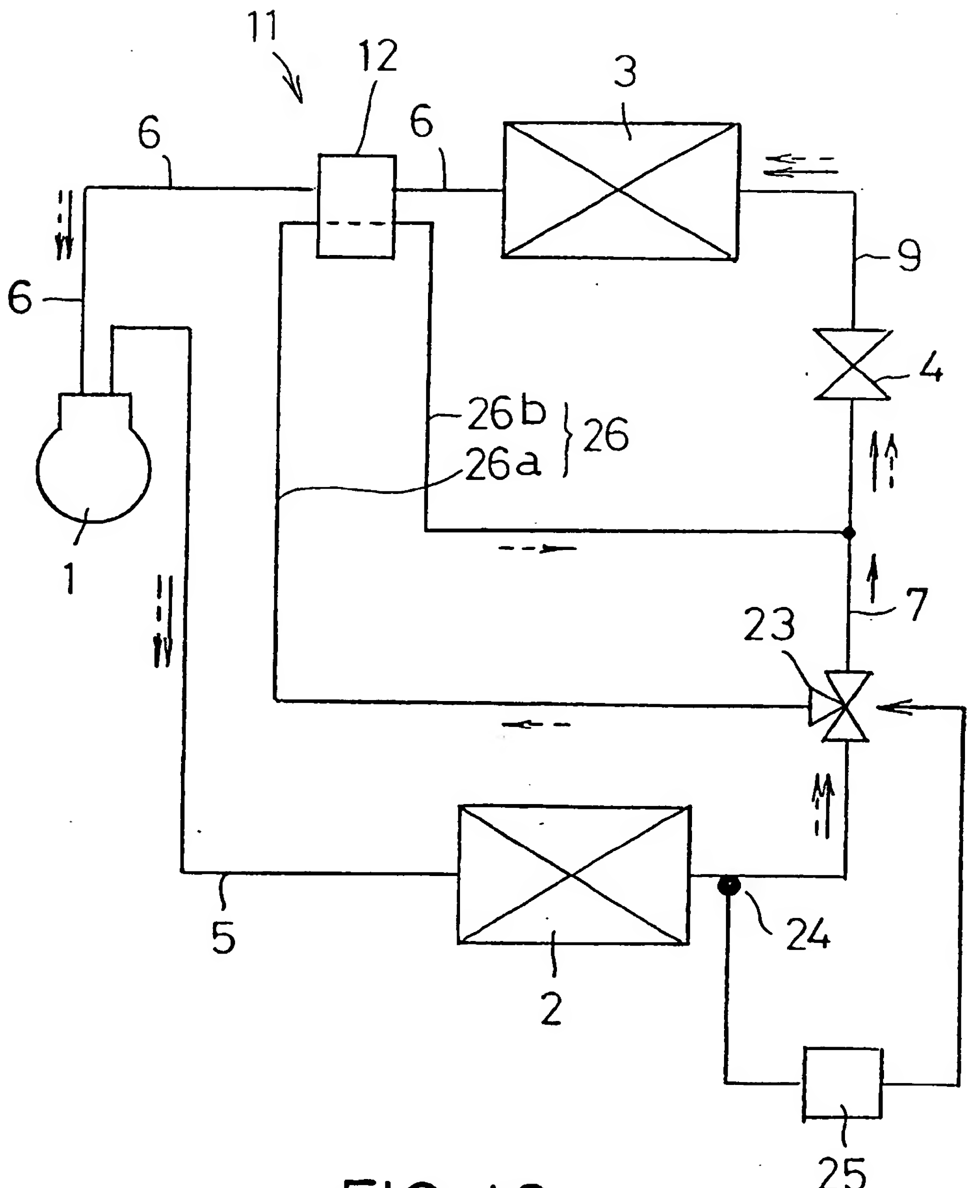

FIG. 12

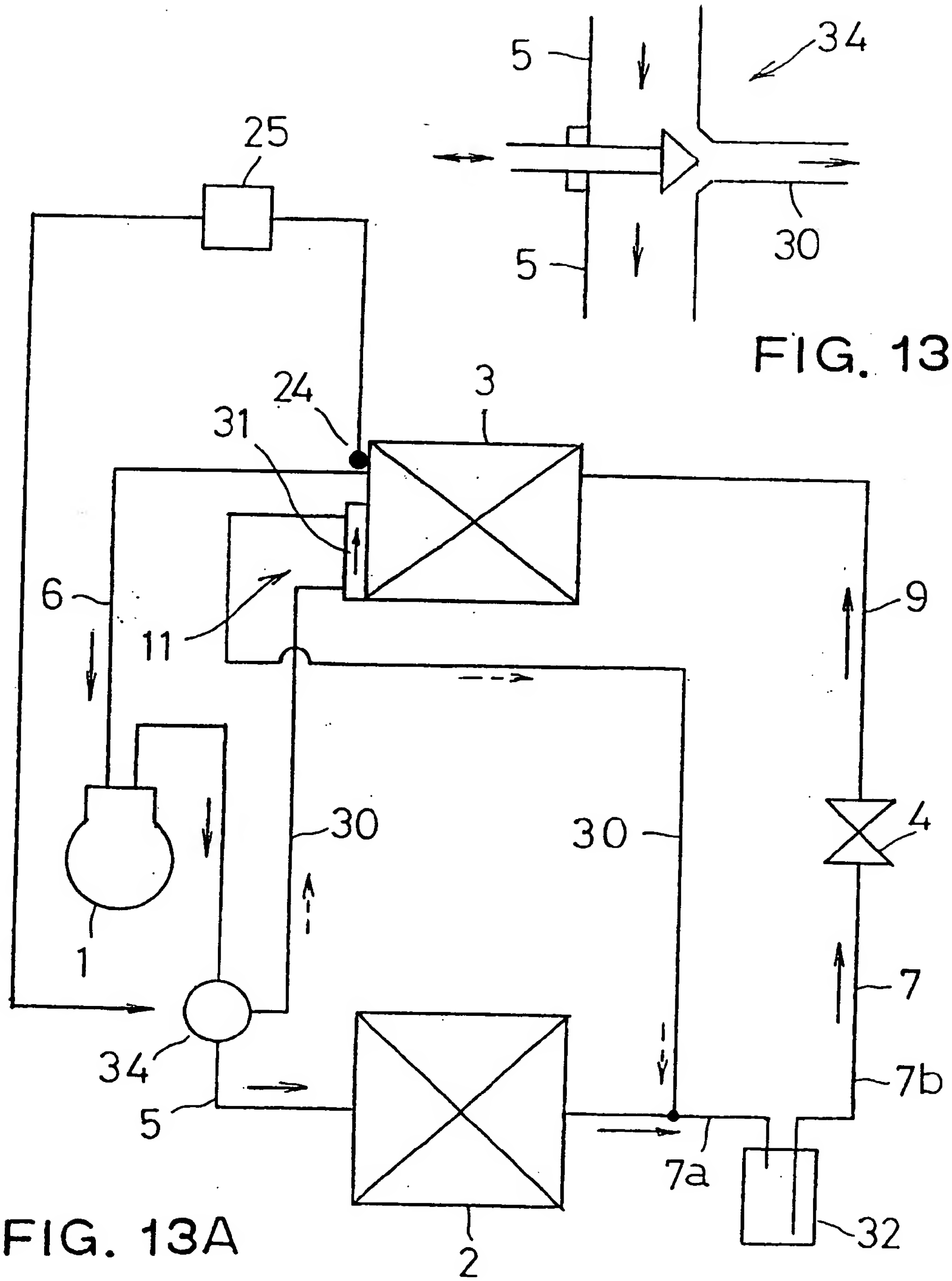

FIG. 13B

FIG. 13A

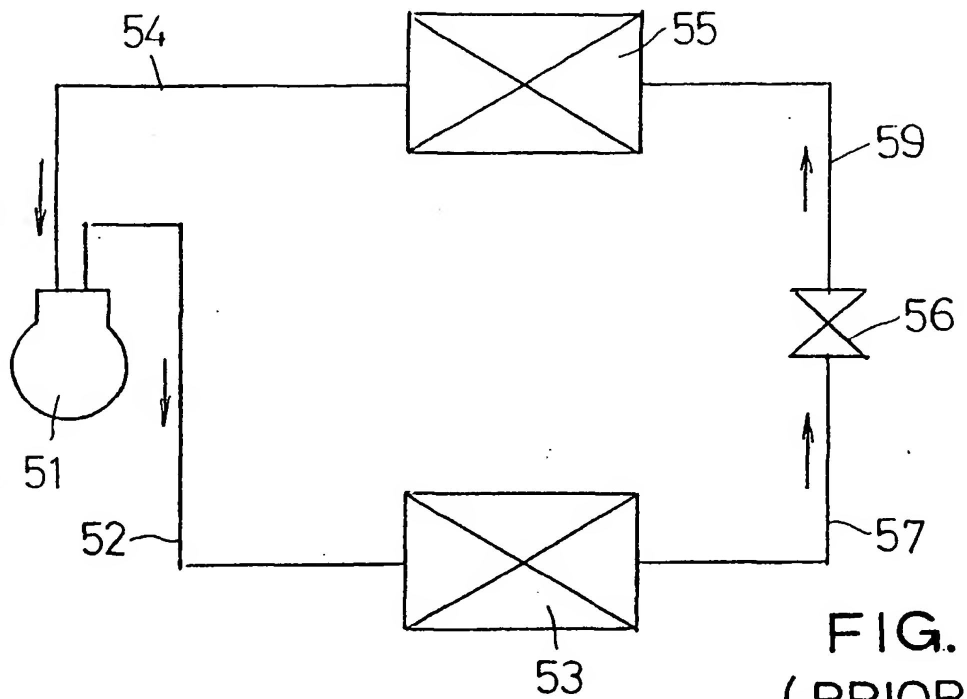

FIG. 14
(PRIOR ART)

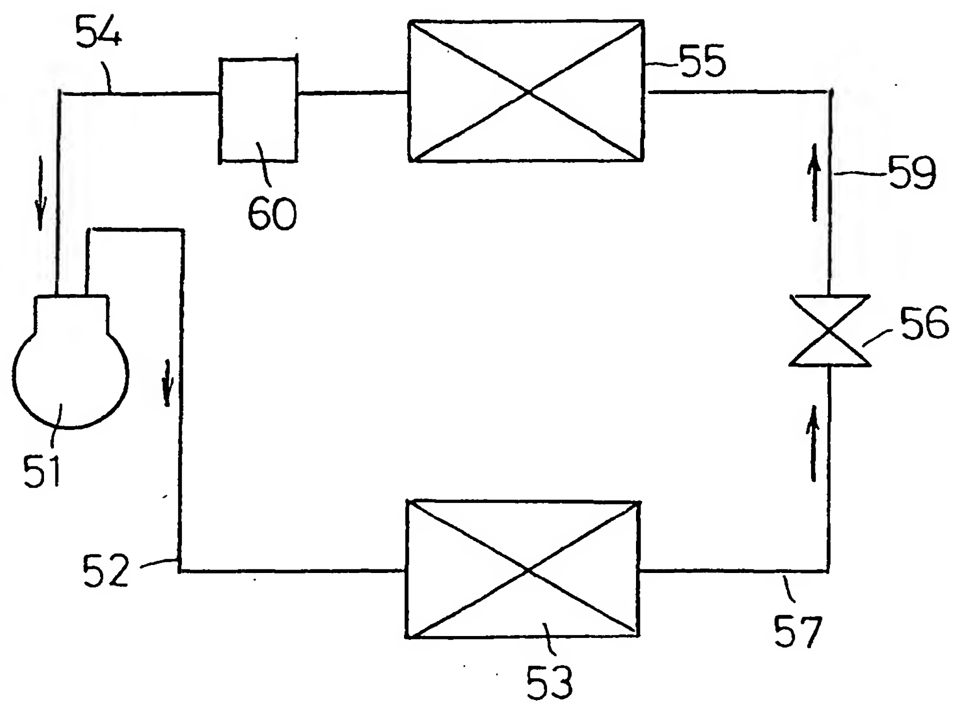

FIG. 15

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(11) EP 0 779 481 A3

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(71) Applicant:
SHOWA ALUMINUM CORPORATION
Sakaishi, Osaka (JP)

(72) Inventors:
- **Nakamura, Junpei,**
  c/o Showa Aluminum Corporation
  Sakaishi, Osaka (JP)
- **Yamazaki, Keiji,**
  c/o Showa Aluminum Corporation
  Sakaishi, Osaka (JP)
- **Higo, Yutaka,**
  c/o Showa Aluminum Corporation
  Sakaishi, Osaka (JP)

(74) Representative:
Paul, Dieter-Alfred, Dipl.-Ing. et al
Fichtestrasse 18
41464 Neuss (DE)

(54) **Refrigeration cycle system**

(57) The present invention relates to a refrigerant cycle system including a compressor (1), a condenser (2), a depressurizing means (4) and a evaporator (3) which are connected in series with each other to form a refrigerant circulation circuit. The refrigerant cycle system includes a heat exchanging portion (11). The heat exchanging portion (11) exchanges heat between a part of or all of the refrigerant passing through a refrigerant passage (5) from the compressor (1) to a depressurizing means (4) by way of the condenser (2) and a part of or all of the refrigerant passing through a refrigerant passage (6) from the depressurizing means (4) to the compressor (1) by way of the evaporator (3).

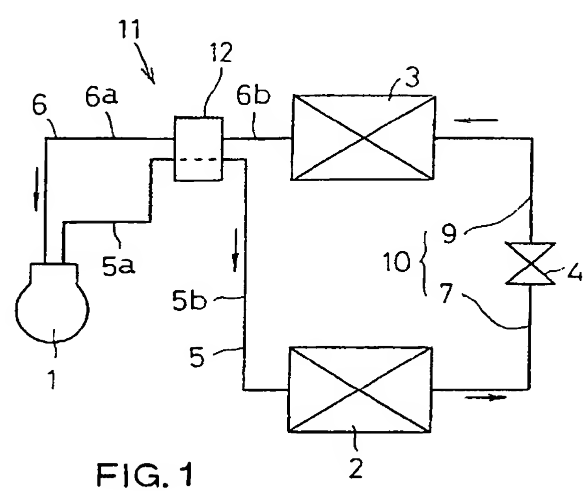

FIG. 1

EP 0 779 481 A3

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 96 11 9908

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X	FR 1 459 402 A (ALLIED CHEMICAL CORPORATION) 6 February 1967 * figures 1,2 * * page 3, column 1, line 38 - page 7, column 1, line 58 * ---	1-4,6, 11,12	F25B40/00 F25B6/02
X	US 4 354 360 A (FISKE HERBERT E) 19 October 1982 * figures 1-11 * * column 2, line 60 - column 10, line 24 * ---	1,2,14	
X	AT 381 787 B (AUSTRIA METALLAKTIENGESELLSCHAFT) 25 November 1986 * figures 1,2 * * page 2, line 28 - line 49 * ---	1,3-5	
X	LU 37 363 A (AMERICAN RADIATOR & STANDARD SANITARY CORPORATION) 27 June 1959 * figures 1,2 * * page 3, paragraph 3 - page 9, paragraph 4 * ---	1,2,6	
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X A	AU 514 567 B (LIQUID MODULATORS INC) 19 February 1981 * figures 1-4 * * page 4, paragraph 8 - page 8, paragraph 3 * ---	1,3,4 5	F25B
X A	US 2 223 900 A (BOWNALL H B) 3 December 1940 * figures 1-8 * * page 2, column 2, line 69 - page 5, column 1, line 30 * ---	1 3,12	
X A	US 2 521 040 A (CASETTA L W) 5 September 1950 * figures 1-9 * * column 1, line 39 - column 4, line 15 * --- -/--	1 10	

~~The present search report has been drawn up for all claims~~

Place of search	Date of completion of the search	Examiner
THE HAGUE	26 January 1999	Nuytens, S

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

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E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03 82 (P04C01)

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 96 11 9908

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X	DE 26 02 529 A (GOETZENBERGER RUDIBERT DIPL IN) 11 August 1977	1	
A	* figures 1,2 *	10	
	* page 7, paragraph 3 *		

X	US 5 245 833 A (MEI VIUNG C ET AL) 21 September 1993	1	
A	* abstract; figure *	14	
	* column 4, line 58 - column 9, line 52 *		

TECHNICAL FIELDS SEARCHED (Int.Cl.6)

~~The present search report has been drawn up for all claims~~

Place of search	Date of completion of the search	Examiner
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Application Number
EP 96 11 9908

CLAIMS INCURRING FEES

The present European patent application comprised at the time of filing more than ten claims.

☐ Only part of the claims have been paid within the prescribed time limit. The present European search report has been drawn up for the first ten claims and for those claims for which claims fees have been paid, namely claim(s):

☐ No claims fees have been paid within the prescribed time limit. The present European search report has been drawn up for the first ten claims.

LACK OF UNITY OF INVENTION

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

see sheet B

☐ All further search fees have been paid within the fixed time limit. The present European search report has been drawn up for all claims.

☐ As all searchable claims could be searched without effort justifying an additional fee, the Search Division did not invite payment of any additional fee.

☐ Only part of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the inventions in respect of which search fees have been paid, namely claims:

☒ None of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the invention first mentioned in the claims, namely claims:

1-6,10-12,14

European Patent Office

LACK OF UNITY OF INVENTION SHEET B

Application Number
EP 96 11 9908

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

1. Claims: 1-6,10-12,14

   Heat exchanging portion in a refrigeration cycle from compressor to condenser, respectively condenser to depressurizing means, and evaporator to compressor.

2. Claims: 1,7,8,9,13

   Heat exchanging portion between bypass from compressor to outlet side of condenser and outermost inner passage of evaporator.

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 96 11 9908

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
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26-01-1999

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EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82